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Design and Performance of an Experimental Cooled Perch System for Heat Stress Relief of Laying Hens

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Abstract

This article summarizes the engineering design and performance of a cooled perch system used in a multi-year collaborative study to evaluate cooled perch effects on hen production, health, and welfare during heat stress. The cooled perch system consisted of two replicates (CP-1 and CP-2) of three-tier cage units with galvanized perch pipes forming a complete loop in each tier (top, middle, bottom) in which chilled water circulated. A total of 324 White Leghorns at 17 weeks of age were randomly assigned to 36 cages (76 cm × 52 cm × 48 cm) in six banks placed in the same room. Flow for each loop was provided by loop pumps that drew chilled water from an open thermal storage manifold and returned it to the same manifold. Each thermal storage was cooled by continuously circulating water through a water chiller. Each loop pump was thermostatically controlled based on the cage air temperature. The water inlet and outlet temperatures, cage air temperatures, and loop water flow rates during stable system operation were measured for performance evaluation. Mean water flow rates in 2015 were 5.19 and 5.45 kg min-1 for CP-1 and CP-2, respectively, but significantly declined to 3.91 and 4.03 kg min-1 in 2016. The mean loop water temperature rise was about 2°C for both replicates. The mean loop net heat gain of CP-1 and CP-2 ranged from 690 to 850 W and from 551 to 1,298 W, respectively, with a significant difference between CP-2 loops ($p < 0.0001$), indicating a discrepancy between the manufacturer's pump curve and field performance. There was a correlation between room air temperature and net heat gain for all loops of CP-1 and the top loop of CP-2 ($p < 0.0001$), suggesting that natural convection and radiation from the room to the pipe were the major contributors to loop heat gain. The average daily net heat gain was approximately 2,334 W per replicate, 256 W m-1 perch length, or 43.2 W per hen housed. This analysis provides a baseline for future cooled perch system design in other application settings. An example is provided for sizing the thermal water storage and chiller capacity. In addition, a closed water system with a properly sized expansion tank is recommended for future energy-efficient cooled perch applications.

Keywords

Alternative cooling, Heat stress, Heat transfer, Poultry

Disciplines

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DESIGN AND PERFORMANCE OF AN EXPERIMENTAL COOLED PERCH SYSTEM FOR HEAT STRESS RELIEF OF LAYING HENS

Y. Xiong, R. S. Gates, J. Y. Hu, P. Y. Hester, H. W. Cheng

HIGHLIGHTS

- This article summarizes the design and performance of a cooled perch system for laying hens during heat stress.
- Water flow rates, perch loop water temperatures, and system net heat gain are provided.
- The average daily net heat absorption data, useful for sizing thermal water storage and chiller capacity in future applications, is 256 W m^{-1} perch length, or 43.2 W per hen housed.
- This analysis provides a baseline for future cooled perch system design in other application settings.

ABSTRACT. *This article summarizes the engineering design and performance of a cooled perch system used in a multi-year collaborative study to evaluate cooled perch effects on hen production, health, and welfare during heat stress. The cooled perch system consisted of two replicates (CP-1 and CP-2) of three-tier cage units with galvanized perch pipes forming a complete loop in each tier (top, middle, bottom) in which chilled water circulated. A total of 324 White Leghorns at 17 weeks of age were randomly assigned to 36 cages (76 cm × 52 cm × 48 cm) in six banks placed in the same room. Flow for each loop was provided by loop pumps that drew chilled water from an open thermal storage manifold and returned it to the same manifold. Each thermal storage was cooled by continuously circulating water through a water chiller. Each loop pump was thermostatically controlled based on the cage air temperature. The water inlet and outlet temperatures, cage air temperatures, and loop water flow rates during stable system operation were measured for performance evaluation. Mean water flow rates in 2015 were 5.19 and 5.45 kg min^{-1} for CP-1 and CP-2, respectively, but significantly declined to 3.91 and 4.03 kg min^{-1} in 2016. The mean loop water temperature rise was about 2°C for both replicates. The mean loop net heat gain of CP-1 and CP-2 ranged from 690 to 850 W and from 551 to 1,298 W, respectively, with a significant difference between CP-2 loops ($p < 0.0001$), indicating a discrepancy between the manufacturer's pump curve and field performance. There was a correlation between room air temperature and net heat gain for all loops of CP-1 and the top loop of CP-2 ($p < 0.0001$), suggesting that natural convection and radiation from the room to the pipe were the major contributors to loop heat gain. The average daily net heat gain was approximately 2,334 W per replicate, 256 W m^{-1} perch length, or 43.2 W per hen housed. This analysis provides a baseline for future cooled perch system design in other application settings. An example is provided for sizing the thermal water storage and chiller capacity. In addition, a closed water system with a properly sized expansion tank is recommended for future energy-efficient cooled perch applications.*

Keywords. *Alternative cooling, Heat stress, Heat transfer, Poultry.*

High ambient temperature, especially acute heat stress events, is one of the most detrimental environmental stressors for the global poultry industry. Heat stress seriously compromises the welfare of laying hens in commercial egg production, negatively affects their performance, and increases mortality, leading to substantial economic losses (Ebeid et al., 2012; Felver-Gant et al., 2012; Lara and Rostagno, 2013; Lay et al., 2011; Mack et al., 2013; Mignon-Grasteau et al., 2015; Zulkifli et al., 2009). According to St-Pierre (2003), hen mortalities climbed to 10% in the Midwest region due to heat events during the summers of 2011 and 2012. A simulation assessment suggested that implementation of "heat abatement" technology would reduce annual economic losses of the U.S. layer industry from $98.1 million to $61.4 million

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by using an economically optimal heat abatement method (St-Pierre et al., 2003).

Several options exist to alleviate the deleterious effects of heat stress. The egg industry regularly alters feed composition with or without additives and provides feed at all times during chronic heat stress periods (Koelkebeck et al., 2014). Physical cooling of the bird or the environment are also options, some of which have become adopted (tunnel ventilation, evaporative cooling), while others have been topics of research (surface wetting, cooled perches). Cooling methods used in the poultry industry include increased air velocity from tunnel ventilation (Botcher et al., 1995) and evaporative cooling systems that use cooling pads and/or fogging nozzles (Bell and Weaver, 2002; Bottcher et al., 1991; Gates and Timmons, 1988; Gates et al., 1991a, 1991b; Timmons and Gates, 1988). Direct wetting of broilers and layers is rarely used due to the challenges of implementation, although investigations have shown some promise (Chepete and Xin, 2000; Ikeguchi and Xin, 2001; Liang et al., 2014; Mutaf et al., 2008; Wolfenson et al., 2001; Yanagi et al., 2002). In addition, several other cooling methods have been examined, including drinking water temperature adjustment (Xin et al., 2002) and cooled perches for broilers and broiler breeders (Muiruri, 1989; Muiruri and Harrison, 1991; Muiruri et al., 1991; Reilly et al., 1991; Zhao et al., 2012).

Cooled perches present an intriguing opportunity for heat removal from birds under heat stress (Muiruri, 1989; Muiruri and Harrison, 1991; Reilly et al., 1991; Zhao et al., 2013). The provision of a cooled perch, in which chilled fluid is circulated through a conventional galvanized perch pipe, offers potential for improved hen welfare and performance during both acute and chronic heat stress events. The method is amenable to both naturally and mechanically ventilated systems, provides a positive welfare aspect by providing birds with a means to express their natural perching behavior, and provides benefits to their skeletal health (Hester et al., 2013; Hester, 2014; Hester et al., 2014; Tactacan et al., 2009). In particular, alternatives to conventional cages, such as enriched colonies and vertical aviaries, provide laying hens with more space and varying environmental features that may be readily modified to incorporate a cooled perch into the design. Cooled perches can provide birds with an alternative means of heat loss via conduction from feet to perch. Because bird legs are highly vascularized, this additional heat loss has potential to offset problems related to compromised ventilation or elevated outside temperature. Previous studies estimated that up to 25% of bird metabolic heat produced can be lost through chicken's legs and feet because of the efficient vascular arrangement (Hillman et al., 1982; Hillman and Scott, 1989). Studies have shown that broiler breeders (parents of broiler meat-type chickens) housed on litter floors with access to cooled perches for three weeks at 35°C ambient temperature had improved egg production, hatchability, and feed consumption as compared to broiler breeders provided with non-cooled or air-equilibrated perches (Muiruri, 1989; Muiruri and Harrison, 1991; Muiruri et al., 1991). Zhao et al. (2013) reported that broiler chickens showed increased weight gain and feed efficiency at high ambient temperatures if provided with cooled perches. However, the effects of cooled perches on laying hen health and welfare have not been widely studied due to challenges with the design and implementation of such systems, including the capital cost, maintenance, and the lack of research data on bird production performance and behavior changes.

We recently conducted a multi-year study using a perch system to examine the effects of water-cooled perches as a cooling alternative on hen performance, plumage condition, foot health, and physiological and behavioral parameters of caged White Leghorn hens exposed to acute and chronic cyclic heat stress events (Cheng et al., 2013; Gates et al., 2014; Hester et al., 2013; Hu et al., 2016, 2019a, 2019b, 2019c). The results for hens housed in the cooled perch (CP) treatment were compared to that of a non-cooled, air perch (AP) treatment, and no perch (control). Information regarding the hen strain, age, experimental design, data collection protocols, and cooled perch effects on hen performance and foot health have been reported previously (Hu et al., 2016, 2019a, 2019b, 2019c).

The results indicated that cooled perches had promising benefits for White Leghorns with regard to performance, plumage condition, foot health, physiological parameters, and post-molt egg production (Hu et al., 2019a, 2019b, 2019c). During chronic cyclic heat episodes, the CP hens had higher egg production ($p < 0.0001$) and feed consumption ($p < 0.04$) than the AP and control hens. The CP hens also had higher body weight at 35 and 72 weeks of age ($p_{treatment\times age} < 0.05$) and a reduced cumulative mortality ($p = 0.02$) compared to control hens but not AP hens. Heavier egg weight ($p < 0.0001$), higher breaking force ($p < 0.0001$), and greater eggshell percentage and thickness ($p_{treatment\times age} < 0.05$) were observed from CP hens than from AP or control hens during heat. No treatment difference was found for nail length, feet hyperkeratosis, and overall feather score (Hu et al., 2019a). For hens aged 21 to 35 weeks (heat stress period 1), CP hens had the lowest rectal temperature ($p = 0.02$) and lower heat shock protein (HSP) 70 ($p = 0.04$) than control hens but not AP hens. For hens aged 73 to 80 weeks (heat stress period 2), CP hens had lower rectal temperature ($p = 0.02$), lower circulating heterophil to lymphocyte (H/L) ratio, and greater packed cell volume than AP hens ($p = 0.02$) but not control hens. The plasma levels of triiodothyronine (T3) and the T3/T4 (thyroxine) ratio for CP hens were higher ($p = 0.002$ and $p = 0.0006$, respectively) than for control hens but not AP hens. No difference was found in cytokines or IgY levels (Hu et al., 2019b). Makagon et al. (2015) reported that for the chronic heat episode, CP hens used the perch at a higher frequency ($p < 0.001$) than AP hens at all observation times. An induced molt study was conducted with the same group of hens after the cyclic heat episode to examine the efficacy of the induced molt on hen production and physiological responses (Hu et al., 2019c). Results showed that at the end of molt, CP hens had higher feed consumption, greater body weight loss, and lower heterophil/lymphocyte ratios ($p < 0.05$). CP hens also had better breast feather scores than AP hens but worse vent plumage ($p < 0.05$).

With the substantial positive benefits provided by the experimental cooled perch system, it is important to document the engineering design and evaluate the system performance for future investigations and perhaps larger-scale application in the poultry industry. The objectives of this study are: (1) to document the design and instrumentation of the cooled perch system; (2) to evaluate perch performance based on water mass flow rate, loop temperature rise, and system net heat gain; and (3) to provide design information for larger-scale application.

Materials and Methods

Cooled Perch System Design

The cooled perch system used in the experiment consisted of a three-tier (top, middle, and bottom) cage unit with two cages per tier, with each cage measuring 76 cm (W) × 52 cm (D) × 48 cm (H). Nine White Leghorn hens were housed in each cage, with approximately 16.8 cm perching space per hen. Two complete replicates were fabricated and designated CP-1 and CP-2. In each CP cage, holes were cut into the center of each end wall to allow passage of two pieces of galvanized perch pipe (33.8 mm O.D. and 28.5 mm I.D.), which functioned as supply and return pipes to form a complete perch loop with two 90° elbows for each tier. Each perch loop was approximately 6.1 m in total length counting all fittings, with 3.0 m of usable length for the hens (Xiong et al., 2015).

Chilled water was separately pumped to each circulating loop on demand from a vertical thermal storage manifold constructed of 13 cm I.D. PVC pipe, 1.70 m tall. Water was returned to the manifold via return lines, which were connected 1.2 m below the supply lines on the manifold. The pump (model 006-B4-15 cartridge circulator, nominal flow of 30 L min^{-1} at 1 m and 43 L min^{-1} at 0 m, Taco Inc., Cranston, R.I.) for each loop was activated when the cage air temperature exceeded the temperature setpoint. The thermal storage manifold was cooled with an independent loop consisting of a fourth pump that continuously circulated water from the manifold through a water chiller (model ER-101y, rated cooling capacity of 0.6 L min^{-1} at 22°C temperature drop, Elkay Manufacturing Co., Oak Brook, Ill.). This water chiller had an independent thermostat set at approximately 10°C. All exposed sections of pipe outside the cages and manifold were insulated with polyethylene pipe insulation (0.033 W m^{-1} K^{-1}) to conserve energy and minimize condensation potential. Figure 1 illustrates the design and instrumentation of the CP system.

The air temperature of the research facility was controlled by a fan ventilation system and hot water heater without evaporative cooling. The single-stage ventilation system had a continuously operating poly-tube distribution system, and

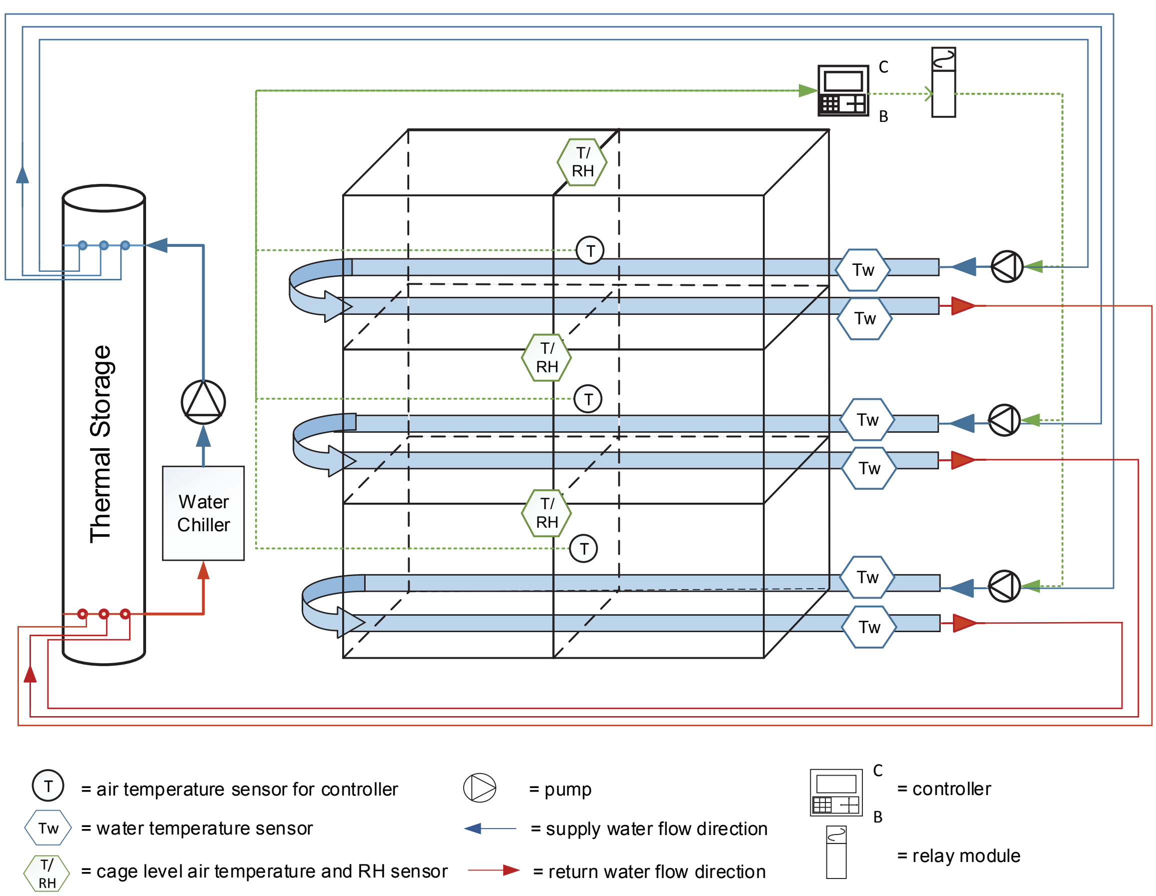

Figure 1. Schematic of cooled perch system and instrumentation. Two systems were fabricated. Each system consisted of three loops (top, middle, bottom) each individually operated by a pump that drew chilled water from a thermal storage manifold and returned to the same manifold. Each pump had effectively the same head loss. The thermal storage was cooled by an independent loop consisting of a fourth pump that continuously circulated water between the manifold and a water chiller. Each loop pump was individually thermostatically controlled based on the air temperature within the cage. Instrumentation included inlet and return line water temperatures, cage air temperatures, and relative humidity (RH).

the room air was well mixed. During the heat stress experiment, the room air temperature was raised to approximately 35°C from 6:00 h to 18:00 h and was stepped down to and maintained at 28°C from 18:00 h to 6:00 h daily.

Environmental Measurements

A wireless data acquisition system (ZW Series, Onset Computer Corp., Bourne, Mass.) was used to monitor and collect ambient and perch loop information. Measurements included cage air temperature (t_{air}) and relative humidity (RH) near the cage ceiling at the partition between the two cages of each tier, and water temperatures of the supply and return pipes of each loop. Room temperature and RH were also collected. All environmental measurements were recorded once per minute.

Thermostatic activation of chilled water to each CP loop (CP-1 and CP-2) was accomplished using a multi-zone controller (CN1514-TH, Omega Engineering, Stamford, Conn.). An air temperature sensor (ON-405, 2252 Ω, Omega Engineering) was installed in each tier, and the controller activated the corresponding circulation pump when t_{air} exceeded the programmed temperature setpoint (25°C). The circulation pump in each tier was activated by a relay module (model URM-400, four electromechanical relays, Omega Engineering) connected via serial interface to the controller. Loop water temperatures were measured using thermistors with a protective cover (TH-44031, Omega Engineering) rated at ±0.5°C.

All sensors deployed in the experiment were checked and, if needed, calibrated within the application range against a National Institute of Standards and Technology certified T/RH device prior to environmental monitoring. An individual calibration equation was applied to each sensor. The environmental measurements were checked daily, and a wireless delivery system was developed to allow remote monitoring.

Water Flow Rate Evaluation

Mass flow rate of the chilled water was used to assess the performance of the cooled perch system and was evaluated using two methods: (1) measured directly, and (2) estimated from empirical equations and the pump curve provided from the manufacturer. The water flow rates obtained from the two methods were compared to determine the system performance uncertainty between system design and field measurement.

Flow Rate Measured Directly

Water flow rate data were collected on a total of seven days over two years during the chronic heat stress experiment (three days in 2015 and four days in 2016), with multiple repeated measurements ($N \geq 3$) taken each day. A tee-valve was installed at the end of each loop where the return water temperature was monitored to create a pathway to capture water without interfering with the experiment. The water stream flowing through each loop for each replicate was collected for approximately 30 s (recorded by stopwatch). The mass flow rate of each loop was calculated by dividing the mass of water captured from the end of each loop by the corresponding elapsed time recorded by stopwatch.

The average water flow rate on each day was used to represent the daily average flow rate. The average water flow rates measured in 2015 and 2016 were taken to represent the respective yearly mean water flow rates for each loop and were plotted for CP-1 and CP-2.

Flow Rate Estimated from Empirical Equations

The mean flow velocity (u_{av}) of the chilled water was computed using equation 1 (Bergman et al., 2011):

$$u_{av} = \frac{\dot{V}}{(\pi / 4) d_i^2} \tag{1}$$

where $\dot{V}$ is the volumetric flow rate estimated from the pump curve, and d_i is the inside diameter (28.5 mm) of the CP pipe.

After obtaining the mean flow velocity of the chilled water, the Reynolds number (Re) was computed by equation 2 (Bergman et al., 2011) and used to determine the characteristics of the chilled water flow circulating in the perch loop:

$$\text{Re} = \frac{\rho_w u_{av} L_c}{\mu_w} = \frac{\rho_w u_{av} d_i}{\mu_w} \tag{2}$$

where ρ_w is the density of the chilled water at 20°C (overall average of water inlet and outlet temperatures across all six loops), ρ_w = 998 kg m^{-3} (ASHRAE, 2017); u_{av} is the mean flow velocity of the chilled water computed from equation 1; L_c is the characteristic dimension for the flow geometry ($L_c = d_i$ = 28.5 mm in this case); and μ_w is the dynamic viscosity of the chilled water, which is 1.003×10^{-3} kg m^{-1} s^{-1} at 20°C.

The pressure drop ($p_1 - p_2$) of the chilled water pumped to the system was calculated by equation 3 (ASHRAE, 2017):

$$p_1 - p_2 = f\left(\frac{\rho u_{av}^2}{2g_c}\right)\left(\frac{L_{eq}}{D}\right) \tag{3}$$

where f is the Darcy-Weisbach friction factor, $\rho u_{av}^2/2g_c$ is the velocity head, and L_{eq} is the equivalent length of the perch loop with all fittings considered. For this analysis, we assumed that the inner surface of the galvanized perch pipe is smooth: $\varepsilon/D = 0.000005$, where ε is the roughness of the pipe surface (mm). The pressure drop that occurs inside the loop in conjunction with the friction factor was estimated by the smooth pipe curve from the Moody diagram (ASHRAE, 2017). The friction losses caused by any fittings present in the loop were converted to equivalent length of pipe. For schedule 40 PVC pipe with a nominal size of 2.5 cm (1 in. PVC), the equivalent length is 0.5 m for elbow ("street el") fittings (Aetna, 2015).

The mass flow rate of the chilled water was estimated from the product of water density and water volumetric flow rate ($\dot{V}$). The newly obtained pressure drop was then used in the pump curve to determine a closer estimate of the volumetric water flow rate. An iterative process was performed by repeating equations 1 to 3 until the pressure drop and the volumetric water flow rate both converged to unchanged values. Four iterations were performed to numerically estimate the final pressure drop and the flow rate of the chilled water. The average measured water flow rate with standard deviation for each loop in each replicate was tabulated for 2015 and 2016. The thermodynamic properties of the chilled water obtained from empirical equations were provided and compared to the measured water flow rate for design uncertainty.

Estimation of System Net Heat Gain

Loop Temperature Rise

The water temperature rise for each loop is useful for evaluating the water chiller performance and estimating the average heat gain of the CP system. Two representative heat stress periods were selected to demonstrate the characteristics of the loop temperature rise. These two consecutive periods contained (1) the water temperature rise profile for 25 to 30 June 2014, during which the two CP replicates performed as expected, and (2) the water temperature rise profile for the following week, 1 to 7 July 2014, during which pump malfunctions were observed. The loop water temperature rise for both CP replicates was calculated from data recorded every minute and was plotted against time during each of the two representative periods. The air temperature of the experiment room was included to indicate the heat stress regime.

Descriptive statistics are provided, including the mean loop temperature rise and standard deviation for the top, middle, and bottom tiers of CP-1 and CP-2. The analysis of loop temperature rise was performed over a period of stable system operation, defined as no sudden changes in loop temperature rise in any tier for either CP replicate, to provide useful information for analyzing system performance as well as designing systems for larger-scale application.

System Net Heat Gain

Obtaining the system heat gain is useful for sizing similar systems proposed for different-scale applications in industry. The descriptive statistics of the measured water flow rate, temperature rise during stable system operation, and the corresponding water characteristics were used to estimate the total heat gain of the CP system. The system net heat gain (W) was estimated using the following equation:

$$Q_{gain} = \dot{m} C_p \Delta t \tag{4}$$

where $\dot{m}$ is the average water flow rate of each loop measured in 2015 (kg s^{-1}), C_p is the specific heat of water at a specific temperature (J kg^{-1} K^{-1}), and Δt is the loop temperature rise calculated at every minute during stable system operation (°C). For this estimation, the specific heat of water at 20°C (the overall average water inlet and outlet temperatures across all six loops) was used. At 20°C, C_p = 4,180 J kg^{-1} K^{-1} (ASHRAE, 2017).

For the representative stable system operation during 25 to 30 June 2014, the system heat gain for each loop in CP-1 and CP-2 was estimated using the water flow rate measured in 2015 and the specific loop temperature rise. The estimated heat gain of each loop was averaged hourly and plotted against time. In addition, the mean water inlet temperatures of the three loops in CP-1 and CP-2 were averaged hourly and plotted against time to demonstrate the water chiller performance. Descriptive statistics, including the overall average net heat gain and the standard deviation over the entire plotted time, are provided for each loop within a replication and for each replication.

Statistical Analysis

The statistical analyses were performed using SAS (ver. 9.4, SAS Institute, Cary, N.C.) for the representative stable system operation period from 25 to 30 June 2014. The hourly net heat gain data from the three tiers in each replication were averaged to represent replicate-level net system heat gain. The hourly net heat gains at both the individual loop level and replicate level for both CP replications were tested for correlation effects with room temperature. The analysis was done using PROC CORR in SAS. PROC UNIVARIATE was used to verify the normality of the dependent variable and accepted at $p > 0.01$. The Spearman method and the Spearman correlation coefficient (r_s) were used to determine correlation effects ($p < 0.05$). Student's t-test was performed using PROC TTEST in SAS for the hourly net heat gain and inlet temperatures of the three loops within each replication to further explore if differences in system heat gain or inlet temperature were present in different loops of the same replication ($p < 0.05$).

Results and Discussion

Results of Water Flow Rate Evaluation

The average water flow rate for each loop from repeated measurements on different days over two years during the chronic heat stress experiment are shown in figure 2. Compared to the measurements taken in 2015, the water flow rates significantly declined in 2016. This decrease in water flow rate was observed for every loop in both CP-1 and CP-2. The average water flow rate for each loop from the three days of measurements in 2015 was 4.85, 5.35, 5.34, 4.97, 5.34, and 6.03 kg min^{-1} for CP-1 top, CP-1 middle, CP-1 bottom, CP-2 top, CP-2 middle, and CP-2 bottom, respectively. These average values decreased to 3.87, 3.97, 3.89, 3.92, 4.05, and 4.12 kg min^{-1}, respectively, in 2016, representing flow rate reductions of 20%, 25%, 27%, 21%, 24%, and 32%, respectively.

The measured water flow rate of the system design was compared to the value estimated using empirical equations (eqs. 1 to 3) and provided an assessment of the uncertainty of the design, which was useful for larger-scale application in commercial egg production. Table 1 lists key heat transfer properties of the CP system achieved by each iteration. The uncertainty of the design was about 77%.

According to the final iterated results estimated from the equations, the Reynolds number was on the order of 10^4, thus the estimated chilled water flow was in the turbulent flow regime. From the Moody diagram (ASHRAE, 2017), the friction factor corresponding to the estimated Reynolds number range for smooth pipe was 0.0275. Subsequently, a 1.2 kPa theoretical final pressure drop was computed for the CP system. A 0.35 L s^{-1} volumetric flow rate of the chilled

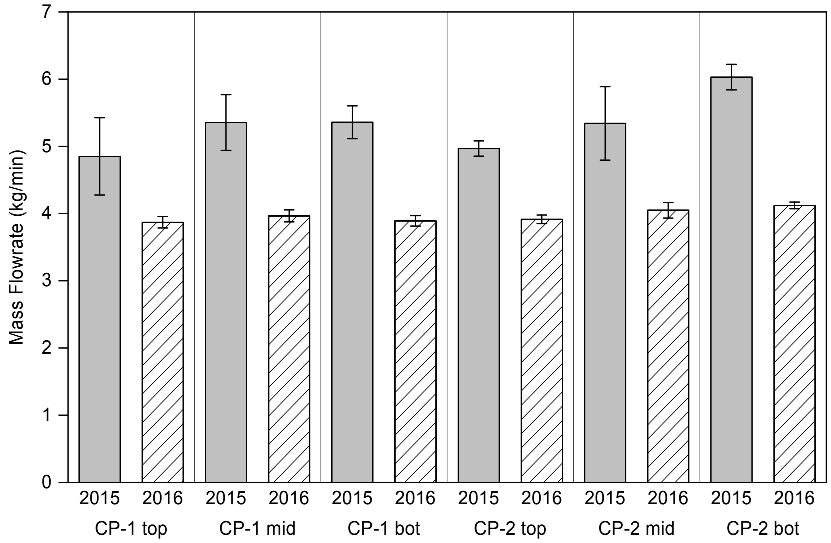

Figure 2. Mean water flow rate for each cooled perch loop in CP-1 and CP-2 measured on a total of seven days over two years during the chronic heat stress experiment. Multiple measurements were taken on each of the seven days, and the average (kg min^{-1}) was taken to represent the water flow rate for each day. Yearly averages of the measurements are shown for 2015 (*N* = 13) and 2016 (*N* = 12) for each loop. Error bars indicate standard deviations of the means.

Table 1. Thermodynamic properties of chilled water in the CP system. Values are means of CP cages during a 24 h heat stress event.

Thermodynamic properties estimated from equations					
Iteration	Reynolds Number (Re)	Friction Factor (f)	Pressure Drop (kPa)	Flow Rate (L s^{-1})	Mass Flow Rate (kg s^{-1})
1	14,738	0.0282	1.0	0.315	0.315
2	16,802	0.0273	1.3	0.359	0.359
3	16,507	0.0274	1.3	0.353	0.353
4	16,359	0.0275	1.2	0.350	0.350
Thermodynamic properties estimated from measured flow rate					
Averaged measured flow rate (kg s^{-1})				0.08	
Reynolds number (Re)				3,776	
Uncertainty of estimation (%)				77.1	

water was calculated using equation 3 and the manufacturer pump curve, resulting in a 0.35 kg s^{-1} flow rate of the system. However, as indicated in table 2, the mean measured water flow rate was only 0.08 kg s^{-1}, which yields a Reynolds number on the order of 10^3 (eq. 2). The discrepancy between the estimated and measured water flow rates generated an uncertainty as high as 77.1%. Based on the range of the Reynolds number calculated from the measured water flow rate, the chilled water flow inside the loops was not in the turbulent flow regime but rather in the transitional flow regime (Bergman et al., 2011). Our traditional understanding of heat transfer processes suggests that such a large discrepancy between calculated and measured water flow rates would likely lead to a corresponding large uncertainty in the heat transfer analysis.

Table 2. Mean and standard deviation (SD) of loop temperature rise for CP-1 and CP-2. The overall mean temperature rise and SD for each CP replicate are included.

	Loop Temperature Rise (°C)			Loop Temperature Rise (°C)	
CP-1	Mean	SD[a]	CP-2	Mean	SD[a]
Top	1.9	0.4	Top	1.7	0.3
Middle	2.1	0.4	Middle	1.2	0.5
Bottom	1.7	0.5	Bottom	3.1	0.2
Overall	1.9	0.2	Overall	2.0	1.0

[a] SD is the standard deviation of the daily means (°C).

Results of Loop Temperature Rise and System Net Heat Gain

Figure 3 shows the loop temperature rise for the two representative heat stress periods. During these two periods, the room temperature followed the same heat stress regime, increasing to approximately 35°C from 6:00 h to 18:00 h and stepping down to 28°C at 18:00 h until 6:00 h the next day. During 25 to 30 June 2014 (fig. 3a), the average (±SD) water temperature rise between the return outlet and supply inlet for each loop was 1.9 ±0.4°C, 2.1 ±0.4°C, 1.7 ±0.5°C, 1.7 ±0.3°C, 1.2 ±0.5°C, and 3.1 ±0.2°C, for CP-1 top through CP-2 bottom, respectively. Based on the theoretical design, a similar water temperature rise should have been observed among all the loops within CP-1 and CP-2. However, this was not observed, as each loop had a different water temperature rise. The overall means of the loop temperature rise were similar between the two replicates, which were 1.9°C and 2.0°C for CP-1 and CP-2, respectively. However, substantial variation in measurements, as indicated by their standard deviations, was observed (CP-1 = 0.2°C and CP-2 = 1.0°C). This indicates that although the same system design, experiment setup, and similar overall temperature differences were noted, the two replicates performed differently. The three loops in CP-1 were more repeatable and stable than those in CP-2.

After the relatively stable system performance period (25 June to 5 July, fig. 3), significant performance changes were observed (fig. 3b), with a rapid disruption in the loop water temperature, as indicated by the supply water temperature being greater than the return water temperature in every loop of CP-2 after noon on 5 July 2014. This trend of negative temperature rise continued for approximately 24 h, after which CP-2 top and CP-2 bottom recovered to their previous

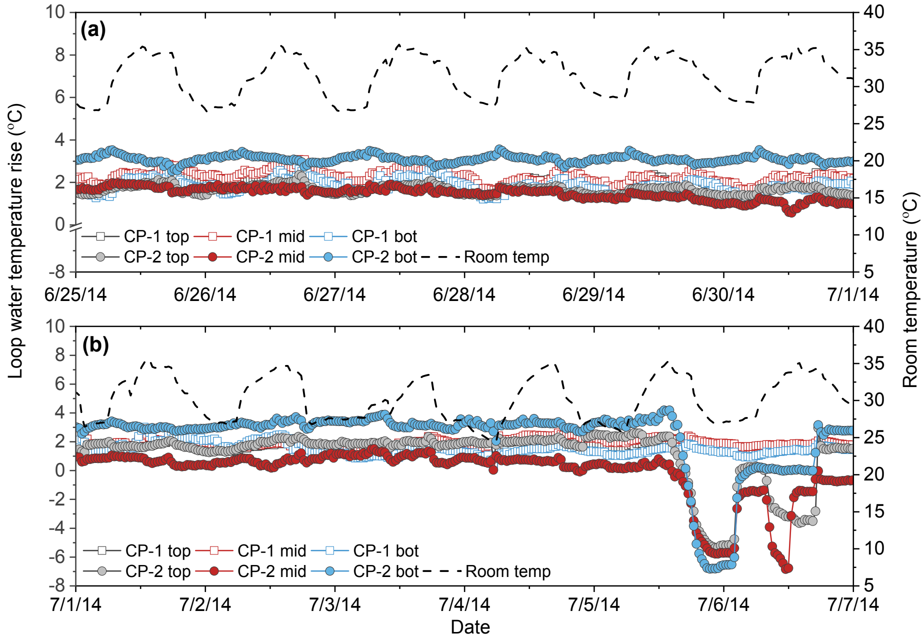

Figure 3. Loop temperature rise for both CP replicates. The temperature rise was calculated as the return water temperature subtracted from the supply water temperature. Data are plotted for heat stress periods (a) 25 to 30 June 2014 and (b) 1 to 7 July 2014, with each symbol representing the temperature difference for each minute. The room air temperature is included for reference. During the plotted periods, the room temperature was increased to approximately 35°C from 6:00 h to 1800 h and stepped down to 28°C from 18:00 h to 6:00 (+1) h daily.

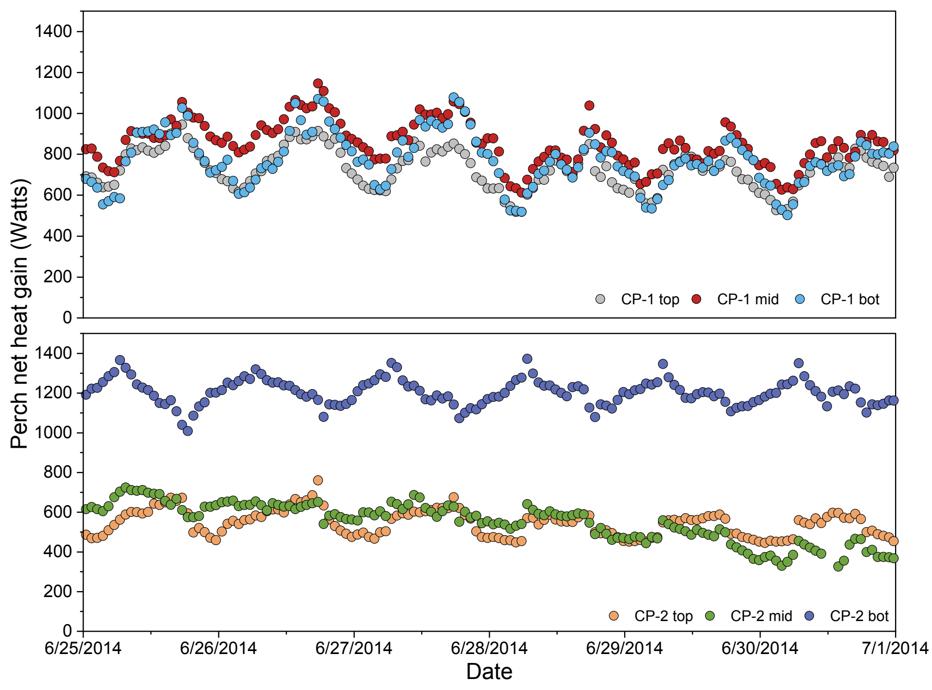

Figure 4. Hourly mean net loop heat gain during stable system operation (25 to 30 June 2014, as shown in fig. 3a). The heat gain was estimated from water flow rate measurements and loop temperature rise. The room temperature increasing to approximately 35°C from 6:00 h to 18:00 h and stepping down to 28°C from 18:00 h to 6:00 h daily.

performance. The unequal performance of the pumps for different loops was also supported by the results from the water flow rate evaluation (fig. 1), in which the water flow rates were different between loops within and across the two CP replicates. Presumably, entrained air in the perch pipes was at least partially responsible for the problem.

Hourly mean net loop heat gains and hourly mean room temperatures during the stable system operation period from 25 to 30 June 2014 are shown in figures 4 and 5, respectively. During this period, room temperature ranged from 26.7°C to 35.5°C, with an average of 31.5 ±2.9°C. Table 3 provides the results of the correlation analysis, including the

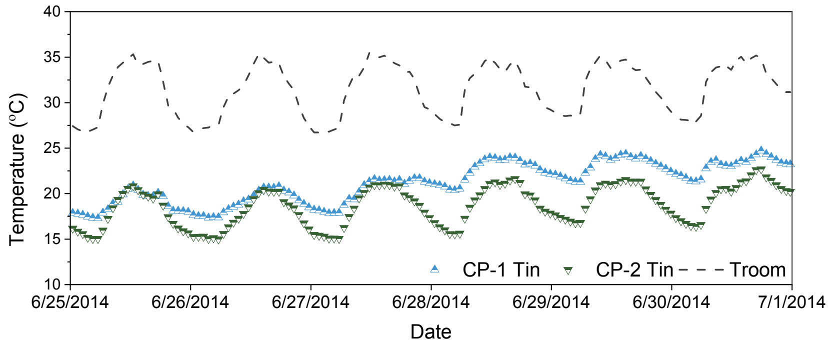

Figure 5. Corresponding to figure 4, mean inlet water temperatures of the three loops in CP-1 (CP-1 T_{in}) and CP-2 (CP-2 T_{in}) and room air temperature (T_{room}) were averaged hourly and plotted against time.

Table 3. Spearman rank correlation coefficients (r_s) between the net heat gain of six perch loops and the room temperature during stable system operation from 25 to 30 June 2014 (N = 144).

Replicate	Perch Level	r_s	p-Value
CP-1	Top	0.73	<0.0001
	Middle	0.43	<0.0001
	Bottom	0.58	<0.0001
CP-2	Top	0.75	<0.0001
	Middle	0.04	0.6031
	bottom	-0.24	0.0042

Spearman rank correlation coefficients (r_s) and p-values between the six net loop heat gains and room temperature. The net loop heat gain for CP-1 was reasonably similar for all three loops and closely paralleled the room air temperature. The net heat gain for CP-1 ranged from 690 to 850 W, with the bottom loop consistently having the lowest net heat gain (690 ±127 W), and the middle loop having the highest net heat gain (850 ±119 W), although without statistical difference ($p > 0.05$). For CP-2, a larger discrepancy was observed for the net loop heat gain at the top and middle levels versus that of the bottom ($p < 0.05$). The average net heat gain of the CP-2 bottom loop (1298 ±78 W) was significantly higher ($p < 0.05$) than that of the top (575 ±74 W) and middle (551 ±112 W) loops.

There was a strong positive correlation between the room temperature and the net heat gain for all three loops of CP-1 ($p < 0.0001$). In other words, the net loop heat gain at the different levels of CP-1 increased with room temperature. During heat stress events, the net loop heat gain for CP-1 peaked daily around noon, when the room temperature also peaked, and declined after midnight, when the room temperature was at minimum. This was also valid for CP-2 top ($p < 0.0001$) but not for the middle and bottom loops of CP-2. There was no correlation for CP-2 middle ($p > 0.05$). There was a weak negative correlation between net heat gain and room temperature ($p < 0.05$, $r_s = -0.24$) for CP-2 bottom, indicating that the heat gain of this perch was slightly decreasing with increasing room temperature. The negative correlation for CP-2 bottom was presumably caused by (1) pump malfunction that did not provide the desired flow rate, (2) air bubbles in the loop that created blockage and prevented water flow, and (3) a coincident operation failure resulting from either or both of the above and controller malfunction that accidentally shut off the circulation pump for CP-2 bottom. This can be partially confirmed by data for subsequent dates starting on 5 July, as shown in figure 3. Hence, the correlation for CP-2 bottom probably does not provide sufficient information for assessing loop heat gain.

System Heat Gain, Chiller Capacity, and Pump Performance

Multiple heat transfer mechanisms drive CP system performance, including (1) heat convection from ambient air to the pipe outside surface, driven by the temperature difference between the air temperature and the pipe outer surface temperature and the outside surface convective coefficient; (2) thermal radiation from the surrounding environment to the pipe outer surface, driven by the difference between the surrounding surface temperature and the radiative coefficient; (3) heat conduction to the perch from hens' footpads, which depends on the effective contact area and the temperature difference between hens' footpads and the perch outer surface; and (4) heat convection of the chilled water inside the perch pipe, which depends on the internal water temperature and the inside surface convective coefficient. The natural convection, radiation, and bird heat conduction should add up to the internal convection to the fluid. The positive correlations (table 3) between room temperature and net heat gain for all levels of CP-1 and for the top loop of CP-2 indicate that natural convection and radiation from the ambient environment to the pipe outer surface were the principle driving force for system total heat gain, compared to bird heat conduction.

The contribution of bird conduction to loop heat gain was probably insignificant compared to convection and radiation from the room, even with the assumption that all birds (N = 18) were on the perch. Albright (1990) provided estimated values of 6.6 W kg^{-1} total heat production and 3.7 W kg^{-1} sensible heat production for a Leghorn laying hen with a typical body weight around 1.8 kg and housed at an air temperature of 28°C. Chepete et al. (2011) estimated a sensible heat production of 3.8 W kg^{-1} for W-36 hens. Given an average body weight of 1.8 kg for laying hens at 30 weeks of age, an average sensible heat production of 6.8 W was a reasonable estimate for this analysis. Hillman et al. (1982) and Hillman and Scott (1989) claimed that a maximum 25% of a bird's total sensible heat production can be transferred from vasomotion through its feet, shank, and bottom leg area. Assuming that all 18 birds were on the perch during the measurements, their maximum potential heat contribution to the loop would be 30.6 W (25% × 6.8 W per hen × 18 hens). In reality, the average number of birds perched on the loop during the day was 11; thus, their actual contribution to loop heat gain was less than 2.5%.

The estimated convection heat gain of each CP replicate using empirical equations (approx. 76.5 W per loop × 3 = 230 W) was only about 1/10 of the system net heat gain calculated using the measured water flow rate and loop temperature difference (2,334 W). A possible hypothesis for this discrepancy is that the loop flow was in the transition phase between laminar and turbulent, which made internal convective heat transfer difficult to predict. The characteristics of transitional flow are difficult to determine, as transitional flow is much more complicated than laminar flow or turbulent flow alone. The literature has demonstrated that natural convection can affect the heat transfer coefficient in the presence of weak forced convection, which may have occurred in this study. As the forced-convection effect increases, "mixed convection" (superimposed forced-on-free convection) gives way to pure forced convection. Grigull et al. (1982) and Metais and Eckert (1964) found that the heat transfer coefficient for a mixed-convection flow regime is often larger than that calculated based on natural or forced convection alone (ASHRAE, 2017).

Although the average inlet water temperatures for the three loops of each replicate exhibited a daily increase (fig. 5), the inlet temperatures rose faster for CP-2 (6°C) than for CP-1 (4°C) from 6:00 a.m. to 1:00 p.m. To further evaluate the system performance, the individual inlet water temperatures of each replicate were analyzed for two representative days during stable operation (28 June and 4 July 2014), as shown in figure 6. Most of the loop inlet temperatures were significantly different from each other ($p < 0.0001$), except for the CP-1 top and middle loops on 28 June 2014 ($p = 0.2049$).

This difference in inlet water temperatures suggests that, despite the use of the same pump models, water chillers, and identical thermal storage manifolds, these components did not perform equally. The pumps used to provide water flow on each tier had marginal performance, without providing enough flow to maintain the designed flow rate, as indicated by the pump performance curve from the manufacturer. As shown in figures 2 and 3, the loop temperature rises and the net loop heat gains were different. If the pumps had provided similar flow rates, then the inlet water temperatures should be similar among different loops because the design provided equal pressure drops for each loop. Given that each pump performed differently, we expected that a lower flow rate would result in a greater loop temperature rise because the pumps did not provide enough flow to push the chilled water through the loop. Similarly, a higher flow rate should result in a smaller loop temperature rise. While this assumption was somewhat supported by the system performance observed for CP-1, it was not the case for CP-2 regarding the relationship between the average measured flow rate and the loop temperature rise. Specifically, as shown in figure 3a, the CP-2 top loop had the second smallest water temperature rise, which suggests that it should have a large flow rate; however, from the measured data, CP-2 top had the lowest flow rate among all six loops. CP-2 bottom presented similar contradictory behavior between the temperature rise and the corresponding average flow rate.

This behavior further indicates that the water chiller and thermal storage manifold were incapable of extracting the total heat transferred to the CP system. The water chiller was rated to provide a 0.0134 kg s^{-1} (12.7 gal h^{-1}) flow rate of 10°C chilled water at 26.7°C (80°F) inlet water temperature and 32.2°C (90°F) room temperature, or a steady-state output of approximately 930 W (eq. 4). However, the 24 h average net heat gain calculated for each replicate was 2,241 for CP-1 and 2,426 W for CP-2, which exceeded the maximum operational cooling capacity by 141% and 161%, respectively. This likely explains the elevated inlet water temperatures, which were 21.1°C and 18.6°C on average for CP-1 and CP-2, respectively. These values are significantly warmer than the chilled water temperature setpoint (approx. 10°C). The system was only able to partially extract the stored heat from the thermal storage manifold, as noted in figure 5. Proper thermal storage sizing is critical to limit the

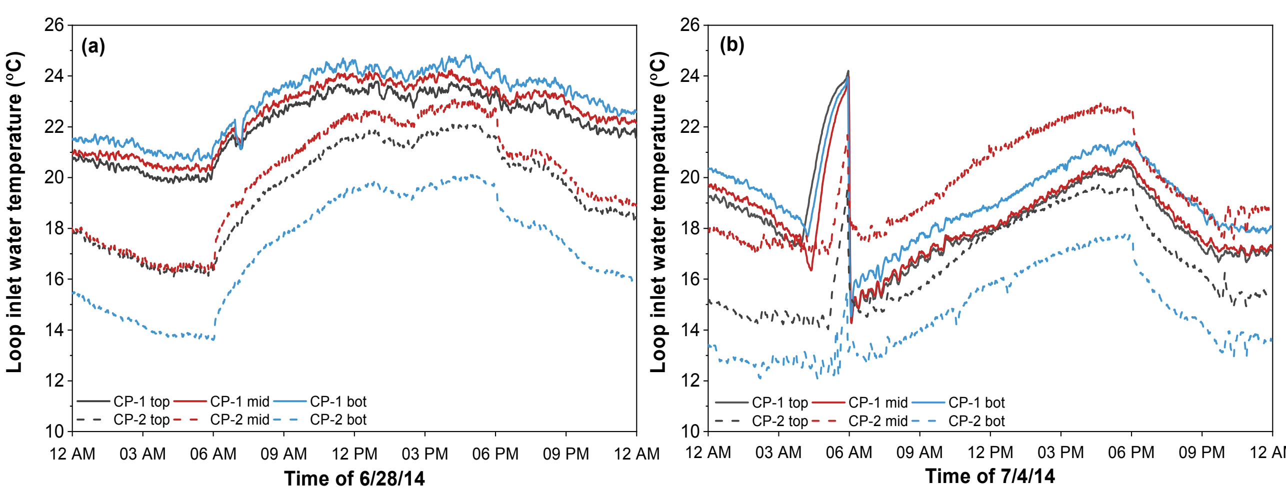

Figure 6. Inlet water temperatures for six loops (CP-1 top through CP-2 bottom) during a 24 h period on two representative days: (a) 28 June 2014 and (b) 4 July 2014. During each period, the room temperature was increased to approximately 35°C from 6:00 h to 18:00 h and stepped down to 28°C from 18:00 h to 6:00 (+1) h daily. The pump for each loop was activated when the air temperature exceeded 25°C.

Table 4. Design criteria for each replicate cooled perch system.

Design Criteria	CP-1	CP-2	Mean
Total average daily net heat gain (W)	2,241	2,426	2,334
Net heat gain per length (W m^{-1})	122.5	132.5	127.5
Net heat gain per hen (W per hen)	41.5	44.9	43.2

loop inlet temperature rise during daily heat stress periods, while avoiding oversized chillers.

Design Criteria

Cooled perch design criteria can be extrapolated from this study. Details for the replicate cooled perches (table 4) provide a basis for selecting chiller and thermal energy storage (TES) capacity using either the full-storage or partial-storage method for a peak-day (ASHRAE, 2016). The average daily net perch heat gain was approximately 2,334 W, or about 256 W m^{-1} perch length or 43.2 W per hen housed. These values are based on the system operating with 12 h day and 12 h night air temperatures of 35°C and 28°C, respectively, and an average loop inlet water temperature of 20°C. Decreasing the inlet water temperature to 10°C with the same temperature schedule would increase the average daily net perch heat gain by approximately 1.9 times. This increase arises from the greater average daily temperature difference between the air and the perch surface (21.5 K vs. 11.5 K).

A more detailed hourly chilled water load schedule is presented in table 5 based on the maximum hourly net perch sensible heat gain for each day and each replicate during stable system operation, and the corresponding partial-storage TES sizing calculation during 25 to 30 June 2014. A chiller output of 2.75 kW, operating continuously, would slightly exceed the peak-day load of 64.4 kW and require a TES capacity of 2.5 kWh and volume of 0.26 m^3 (the TES volume is computed from equation 4 in chapter 51 of ASHRAE (2016) using 90% efficiency and a 9 K temperature difference; actual values depend on the application and tank configuration). These calculations suggest that the experimental CP system, with approximately 900 W chiller output and 0.02 m^3 TES volume, was undersized.

Opportunities for energy savings and trade-offs between chiller capacity and TES volume could be exploited. For example, for the daily load profile in table 5 for this system, shutting down the loop pumps for 8 h each night (i.e., daily total reduced from 64.4 to 44.8 kW), combined with not running the chiller for 6 h during the afternoon peak electricity period, could result in energy savings with the same chiller capacity. In this case, the TES volume must increase from 0.26 to 2.1 m3. In this example, the capital cost of the added TES volume must be balanced against the chiller power costs.

Challenges and Recommendations

Air bubble formation in the perch loops was a persistent challenge throughout the study and impacted the ability to maintain desired system operation. Symptoms of air bubble formation and blockage included steadily increasing loop outlet temperatures combined with relatively elevated and stable loop inlet temperatures. In addition, air bubbles were occasionally seen being purged into the thermal storage manifold. This may have been caused by the open-top design of the storage manifold, as compared to a closed system. To prevent air blockage in potential future research or larger-scale application, we recommend a closed system to avoid system flow change due to elevation change. A closed system should have a properly sized expansion tank that is suitable to maintain a constant system pressure during operation, with an air/water separator. A water flowmeter on each loop would be useful for diagnosis and validation of system performance.

Table 5. Peak-day full-storage thermal energy storage sizing calculation for a stratified chilled water (CHW) system.

	Cooled Perch Hourly Sensible Heat Gain for Each Replicate (kW)													Chiller	TES
	25 June		26 June		27 June		28 June		29 June		30 June		Q_{max}	Output	Charge
Time	CP-1	CP-2	CP-1	CP-2	CP-1	CP-2	CP-1	CP-2	CP-1	CP-2	CP-1	CP-2	(kW)	(kW)	(kWh)
12:00 a.m.	2.20	2.29	2.27	2.37	2.30	2.26	2.28	2.19	2.06	2.11	2.02	2.00	2.37	2.75	0.44
1:00	2.18	2.32	2.33	2.45	2.25	2.34	2.16	2.20	2.13	2.15	1.96	2.03	2.45	2.75	0.74
2:00	2.08	2.31	2.14	2.45	2.20	2.32	1.83	2.20	1.85	2.15	1.74	2.01	2.45	2.75	1.04
3:00	1.93	2.34	2.03	2.43	2.05	2.31	1.71	2.21	1.76	2.14	1.68	2.03	2.43	2.75	1.36
4:00	1.93	2.42	2.07	2.48	2.03	2.40	1.68	2.25	1.80	2.19	1.67	2.05	2.48	2.75	1.63
5:00	1.95	2.52	2.14	2.47	2.05	2.37	1.65	2.27	1.87	2.20	1.75	2.11	2.52	2.75	1.86
6:00	2.07	2.63	2.28	2.56	2.29	2.57	1.89	2.58	2.20	2.45	2.01	2.37	2.63	2.75	1.98
7:00	2.43	2.64	2.39	2.51	2.43	2.55	2.00	2.48	2.23	2.38	2.10	2.27	2.64	2.75	2.09
8:00	2.55	2.61	2.43	2.48	2.54	2.47	2.15	2.38	2.32	2.33	2.25	2.20	2.61	2.75	2.23
9:00	2.63	2.55	2.41	2.51	2.43	2.46	2.18	2.38	2.39	2.29	2.38	2.18	2.63	2.75	2.35
10:00	2.64	2.53	2.48	2.50	2.64	2.53	2.30	2.43	2.40	2.25	2.37	2.12	2.64	2.75	2.46
11:00	2.63	2.51	2.63	2.46	2.83	2.48	2.39	2.37	2.31	2.22	2.24	1.99	2.83	2.75	2.38
12:00 p.m.	2.61	2.52	2.83	2.51	2.69	2.39	2.29	2.34	2.30	2.27	2.32	2.05	2.83	2.75	2.30
1:00	2.60	2.48	3.02	2.50	2.77	2.38	2.22	2.31	2.21	2.27	2.40	2.14	3.02	2.75	2.03
2:00	2.69	2.45	2.88	2.47	2.77	2.39	2.10	2.36	2.33	2.27	2.25	2.12	2.88	2.75	1.90
3:00	2.76	2.47	2.79	2.48	2.72	2.40	2.27	2.40	2.24	2.24	2.22	2.24	2.79	2.75	1.85
4:00	2.77	2.43	2.83	2.52	2.78	2.44	2.56	2.40	2.31	2.30	2.43	2.28	2.83	2.75	1.77
5:00	3.03	2.32	3.13	2.58	2.99	2.44	2.79	2.26	2.59	2.22	2.58	2.17	3.13	2.75	1.39
6:00	2.87	2.18	3.05	2.25	2.93	2.25	2.49	2.07	2.58	2.04	2.47	2.00	3.05	2.75	1.09
7:00	2.66	2.16	2.83	2.29	2.82	2.29	2.31	2.16	2.46	2.04	2.46	2.06	2.83	2.75	1.00
8:00	2.58	2.23	2.80	2.26	2.66	2.26	2.45	2.14	2.35	2.02	2.41	2.00	2.80	2.75	0.00
9:00	2.46	2.28	2.64	2.22	2.41	2.20	2.33	2.08	2.30	1.99	2.41	2.00	2.64	2.75	0.11
10:00	2.32	2.30	2.47	2.20	2.33	2.16	2.17	2.11	2.18	1.98	2.30	2.01	2.47	2.75	0.47
11:00	2.30	2.30	2.39	2.20	2.31	2.19	2.12	2.13	2.04	1.97	2.39	1.98	2.39	2.75	0.82
												Total	64.4	66.0	

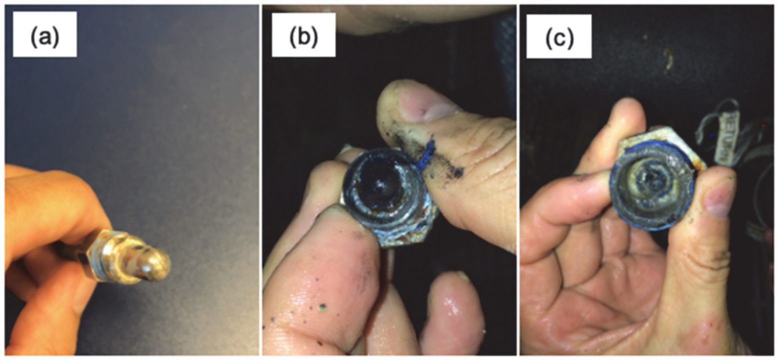

Figure 7. Sensor condition: (a) clean sensor prior to installation, and (b) sensor and (c) sensor housing after one year of installation.

Despite regular treatment with quaternary ammonia, biofilm buildup inside the system was observed throughout the study. Biofilm accumulated inside the cooled perch loops and on the inlet and outlet water temperature sensors (fig. 7). Biofilm accumulation may have reduced the response time of the water temperature sensors and affected pump performance and lifespan. By using a closed system, the amount and frequency of biofilm buildup is expected to be at minimum.

Conclusions

Positive benefits to the laying hens in terms of physiology, foot health, and performance were realized by the experimental cooled perch design. A strong correlation was noted between room temperature and perch net heat gain, indicating that natural convection and radiation from the room air and surfaces to the pipe outer surface were the major contributors to heat gain over other heat transfer mechanisms, including conducted bird heat. The cooling system improved hen heat tolerance, although different loops and replicates did not have equal performance regarding water flow rate, loop water temperature rise, and loop net heat gain. Average daily heat gain was about 256 W m^{-1} perch length or 43.2 W per hen housed, based on 12 h day and 12 h night air temperatures of 35°C and 28°C, respectively, and an average loop inlet water temperature of 20°C. A peak-day system heat load of 64.4 kW was estimated, requiring a thermal storage capacity of 2.5 kWh (equivalent to 2.1 m^3 volume). The results warrant future studies of CP application with a closed system.

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